THERMODYNAMIC ASSESSMENT OF PLANT EFFICIENCIES FOR HTR POWER CONVERSION SYSTEMS

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We studied thermodynamic aspects influencing the efficiency of 200 MWth-HTR modules, considering steam cycle plants, gas turbine plants with and without intermediate heat exchanger and plants with combined gas turbine/steam cycle. The influence of steam parameters and of reheating is investigated. The investigated parameters with major influence on the efficiencies of gas turbine cycles are the core outlet temperature, polytropic efficiency of the turbomachinery, performance of the recuperator and the cooling temperature. For combined cycles, we have investigated a direct combined cycle with gas turbine and steam generator heated by the exhaust gas of the turbine, and a combined cycle with intermediate heat exchanger. For core outlet temperatures of 900 ^{0}C, and steam temperatures of 600 ^{0}C, respectively, cycle efficiencies between 40.7% and 47.5% have been found, with the indirect gas turbine cycle on the lower bound and the direct combined cycle on the upper bound of the spectrum.

1. Introduction

Conventional power plants – steam cycle, gas turbine and combined cycle plants – are improved continuously. As the HTR can principally be coupled with a variety of those power conversion systems, it was proximate to study the related potential also for the HTR.

Historically, HTR plants have or had steam cycles mainly because they are close to conventional steam plants. New conventional plants have steam temperatures of 600...700°C, for improving the plant efficiency. For an HTR, there are no principal obstacles to be in line with the actual trend, i.e. generate steam with similar temperatures.

Gas turbine cycles are regarded to be attractive for the HTR, mainly because of their potential for a high thermodynamic efficiency. Considerable effort had been invested in the design and experimental R&D for gas turbine cycles, but up to the present such plants were not realized. In the meantime, the progress of gas turbines has stimulated new interest for the gas turbine cycle for the HTR.

Combined cycle plants which connect the gas turbine and the steam turbine have gained increasing importance in conventional power plants. Therefore, we found it reasonable to study the application of this technology for HTR power conversion systems, too.

In our analysis we considered, with restriction to thermodynamic aspects, steam cycles with and without reheater, gas turbine cycles with and without intermediate heat exchanger (IHX), and combined cycles with and without IHX.

2. Main assumptions

The following important parameters and main assumptions have been used in our analyses. <u>Reactor power and core outlet temperature:</u> We set the reactor thermal power to 200 MW and we assumed, that a reactor design combining this power and a gas outlet temperature of 900°C can be realized which is compatible with the 1600°C-concept. It requires, that peak core temperatures must not exceed this value in a core heat-up accident.

Temperature difference in the IHX: In the former German nuclear process heat project the design provided an IHX between the reactor and the coal gasifier, with inlet/outlet temperatures of 950/293°C on the primary side and 220/900°C on the secondary side, respectively. This temperature difference in the IHX of 50 K has been experimentally confirmed and we took it as reference value.

Polytropic efficiency of the gas turbomachinery: For optimized gas turbines and compressors, the polytropic efficiency is in the order of 85% to 89%. We set the polytropic efficiency to 85% since the HTR turbine is smaller than conventional large turbines, and the working fluid is helium.

Temperature difference in the recuperator: The performance of the recuperator can be reflected by the temperature difference between the hot and the cold side. In the industrial recuperator design of the HHT project, the average difference was about 25 K, and we selected it as the reference value.

Cooling temperatures in the coolers in a gas turbine cycle: For the coolers, their cooling temperatures will be determined by the state of the art of the coolers and the environment. In the designs of the GT-MHR [1], the PBMR [2] and the MPBR [3], the cooling temperatures are 26, 27 and 30°C respectively. We took 30°C as a conservative reference value.

Live steam conditions: In advanced coal-fired steam power plants, supercritical steam parameters (up to 300 bar and 600°C) are applied. For the HTR steam cycle, we have chosen a live steam temperature of 600°C as a reference, and a live steam pressure of 160 bar for the no reheating steam cycle and 250 bar for the reheating steam cycle, respectively.

Condenser pressure and quality of the exhaust steam: Low condenser pressures (0.03 – 0.04 bar) are desired, but their achievement depends on the site condition. We set the condenser pressure to 0.05 bar in our analyses, typical for wet cooling towers. Normally the quality of the exhaust steam should be higher than 85%. For the German light water reactor convoy plants, the wetness of steam in the low-pressure turbine can be as high as 14%. Thus, in our analyses, 86% has been set as the lower baseline.

Isentropic efficiency of the steam turbine: The isentropic efficiency of an optimized steam turbine is 90% for an advanced power plant. Conservatively, we took a value of 87% in our analyses.

Efficiency of the feedwater pump: According to manufacturers informations, the overall efficiency of the feedwater pump can be 75-85% according to the size. So, we set the efficiency to 75%.

Cycle efficiency and cycle consumption: The cycle efficiency denotes the ratio of the net electricity production, which equals the difference between the generator gross production and the power consumed by the cycle itself to the total nuclear thermal power. In our definition of the cycle consumption, we took only the power use of the helium blower and feedwater pumps into consideration.

3. Results

3.1. Steam cycles

The plant efficiencies of HTR modules with one reheater and without reheater have been calculated. The main thermodynamic conditions can be found in Table 1. The steam cycle without reheater yields, for live steam conditions of 160 bar/600 ^{0}C, a plant efficiency – ac-cording to our definition – of 41,5%. In the process with one reheater, live steam of 250 bar/600 ^{0}C is expanded in the high pressure turbine section down to 85 bar. Then it is heated up again in the reheater to 600 ^{0}C and led to the middle/low pressure section of the turbine. In this case, the efficiency yields 44,3%. The depen-dence on the live steam temperature and pressure is depicted in Fig. 1. For a 1 percentage point gain in effici-ency, the live steam temperature must be elevated by about 60 K, or the pressure by about 50 bar. The figure shows also efficiencies for 177,5 bar /530 ^{0}C, the values of the

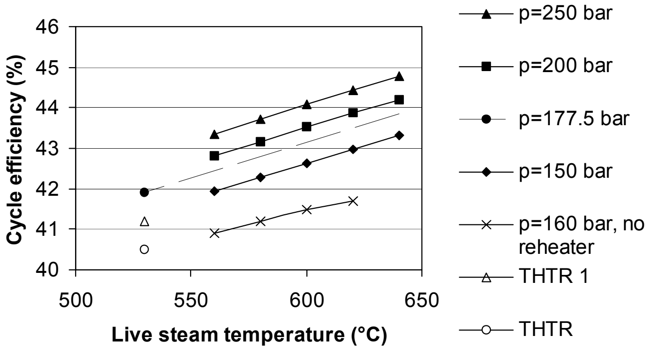

Fig. 1 Cycle efficiencies for steam cycles with and without one reheater

THTR which had one reheater. The point ‘THTR’ denotes the efficiency as has been published [4]. ‘THTR1’ denotes our calculation, assuming isentropic efficiencies of 87% for all three sections of the turbine, and a condenser pressure of 0,0685 bar. This was valid for the THTR which had dry air cooling. The difference of about 0,5% between our calculation and the THTR-value is low and can be attributed to some parameters which are not known to us, e.g. the true efficiencies of the three turbomachine sections. For the calculation of the upper point, the condenser pressure was 0,05 bar, all other assumptions equal to point ‘THTR1’.

3.2. Gas turbine cycles

The gas turbine cycle has the thermodynamic advantage that it can make direct use of the high upper process temperature of an HTR. We calculated a direct cycle consisting of core, high pressure-, low pressure- and power turbine, recuperator, precooler, low pressure compressor, intercooler, and high pressure compressor. The core outlet temperature affects the plant efficiency significantly, see Fig. 2.

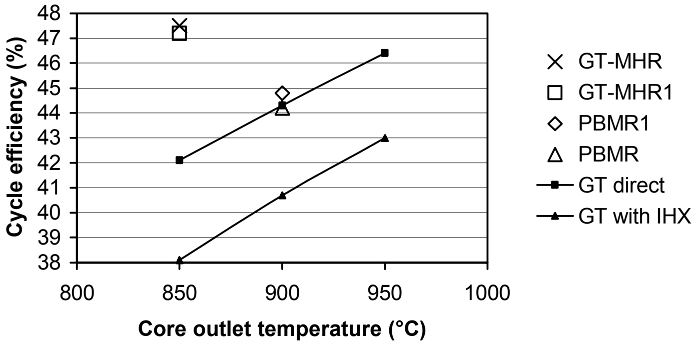

Fig. 2 Cycle efficiency vs. core outlet temperature

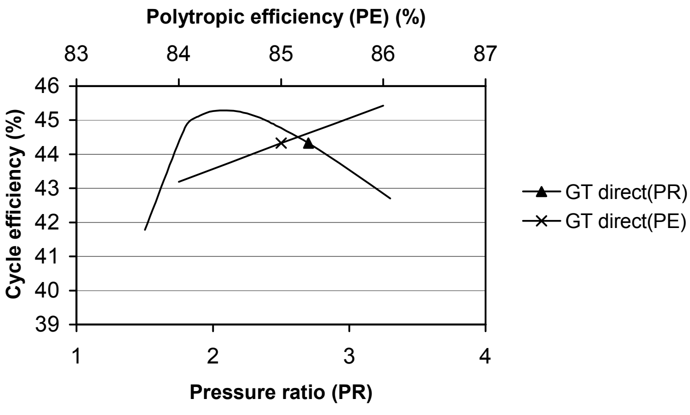

Fig. 3 Cycle efficiency vs. pressure ratio and polytropic efficiency of the turbomachinery for direct gas turbine cycles

When the pressure ratio is fixed and the core outlet temperature is raised by 50 K, the plant efficiency raises by about 2 percentage points. The point for GT-MHR is taken from [1], and point GT-MHR1 shows the result of our calculation with the same assumptions. For the GT-MHR, the cooling temperature of 26 ^{0}C is very favorable, and the polytropic efficiency of the turbomachinery is high. For the PBMR [2], the cooling temperature is 27 ^{0}C, and with a nuclear thermal power of 265 MW and a generator production of 117 MW the efficiency according to our definition is 44,2%. Based on the design parameters of the PBMR, our calculation gives an efficiency of 44,8%, see point PBMR1 in Fig.2. This agrees sufficiently with the value given in the literature.

Other parameters, besides the core outlet temperature, which can influence strongly the cycle efficiency are the pressure ratio, the polytropic efficiency of the turbomachinery, the performance of the recuperator and the cooling temperature. The cycle efficiency vs. pressure ratio and polytropic efficiency of the turbomachinery is illustrated in Fig. 3. There are however restrictions to use the optimal pressure ratio of around 2,1. With a low pressure ratio the core inlet temperature raises, and if e.g. with respect to the pressure vessel the core inlet temperature must be below 560 ^{0}C, a pressure ratio of 2,7 must be taken. The cycle efficiency increases 1 percentage point if the turbomachinery efficiency is elevated by 1 percentage point. If the temperature difference in the recuperator is reduced from 25 K to 20 K, then the cycle efficiency will be increased by 0.7 percentage points. The cycle efficiency will deviate by about 0.5 percentage points if the cooling temperature changes by 3 K.

The main advantage of gas turbine cycles with IHX is the fact, that the whole complicated power conversion unit is clearly separated from the primary circuit, contamination free and with easy access to all components. However, there is a loss of 50 K in the upper process temperature of the secondary cycle. The blower in the primary helium circuit can only be operated up to certain temperature limits. We have chosen 450 ^{0}C as the upper value. As the core inlet temperature is also determined indirectly via the IHX by the pressure ratio in the secondary cicuit, the pressure ratio there must be as high as 4.5, which is much higher than the optimal value. Cycle efficiencies for gas turbine cycles with IHX are depicted also in Fig. 2.

3.3 Combined cycles with gas turbine and steam turbine

The combination of gas turbine and steam turbine leads to the most efficient conventional power plant systems which can reach electric efficiencies next to 58%. Principally, combined cycles are feasible for the HTR, too. From the great variety of possibilities we have selected two cycles: a closed, direct combined cycle and an open combined cycle with IHX.

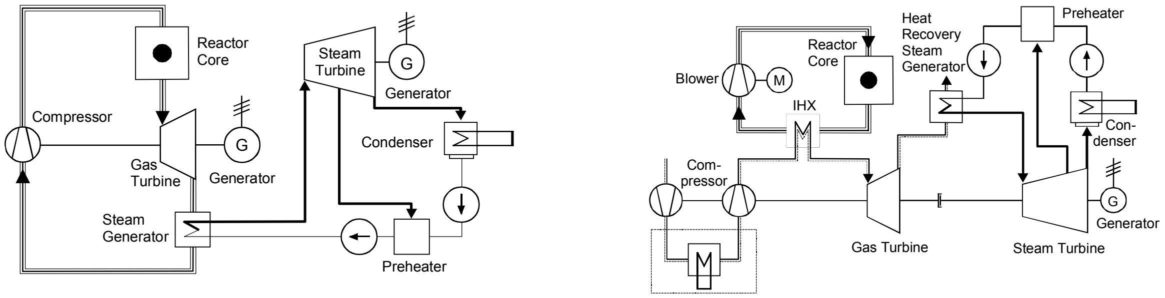

Fig. 4 Combined cycles: direct (above) open with IHX

The direct cycle provides maximum efficiency. The primary circuit consists of the core, a gas turbine, a heat recovery steam generator and a compressor (see Fig. 4). The gas turbine exhaust of 600°C is used for steam generation. With a temperature difference between the off-gas and the steam side of 50 K, a live steam temperature of 550°C can be reached. This cycle obtains an efficiency of 47.5% for 900 ^{0}C core outlet temperature. Within this calculations two aspects have to be considered. On one hand there is the optimum pressure ratio to achieve maximum efficiency. On the other hand there are limits which are affected by the choice of the pressure ratio, e.g. the steam temperature and the operating temperature of the blower. Respecting these aspects, the resulting dependence of the cycle on the reactor temperature is shown in Fig. 5.

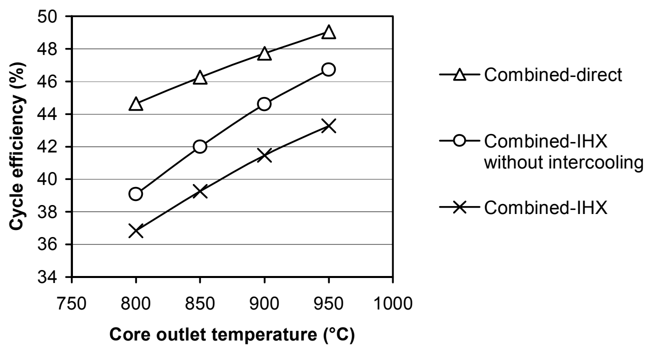

Fig.5 Combined cycles: efficiencies vs. core outlet temperature

An IHX together with a combined cycle has similar advantages and disadvantages, as dicussed in chapter 3.2. Fig. 5 shows the dependence of efficiency on core outlet temperature for two versions of a combined cycle with IHX: One version operates with intercooling between the compressor stages, yielding 41.5% for 900 ^{0}C core outlet tempera-ture. A higher efficiency can be reached by omitting the intercooling. Intercooling has two main effects. Firstly, an intercooler reduces the reactor inlet temperature. Without intercooler, given limits may be exceeded, e.g. for the blower. Also, an intercooler leads to a lower medium compression temperature, which causes less compression power. Secondly, the intercooler rejects heat from the process. For the considered combined cycles with IHX the last aspect has the biggest impact on efficiency. The efficiency without intercooling yields 44.6%.

4. Comparative overview

The potential to improve the efficiency by lifting the reactor core outlet temperature or steam temperature, respectively, is shown in Fig. 6, combining the results of the preceeding sections. All cycles comprising a gas turbine show a similar increase in efficiency if the core outlet temperature is lifted, on different levels for the different processes. The same tendency can be observed for steam cycles, if the live steam temperature is lifted. Fig. 6 shows three groups of cycles, arranged according to the degree of complexity and proof of technology. The cycles with the lowest efficiency, denoted A in Fig. 6, have moderate complexity and predominantly proven technology. Better efficiency is achieved with higher complexity or not nuclear-proven technology, B, and the best theoretical values are achieved with high complexity and not nuclear-proven technology, C. Certainly the core outlet/steam temperatures are by far not the only means to improve the efficiency. Other parameters like the isentropic efficiency of the turbines or of the recuperator are are also of great influence. The calculated efficiencies must be regarded as first-order approximations. It is often observed, that efficiencies „degrade“ in parallel to the progress of the design in a real construction project. The overview on efficiencies should stimulate a more comprehensive assessment on power cycles, which must include such important questions like investment costs, or development risks.

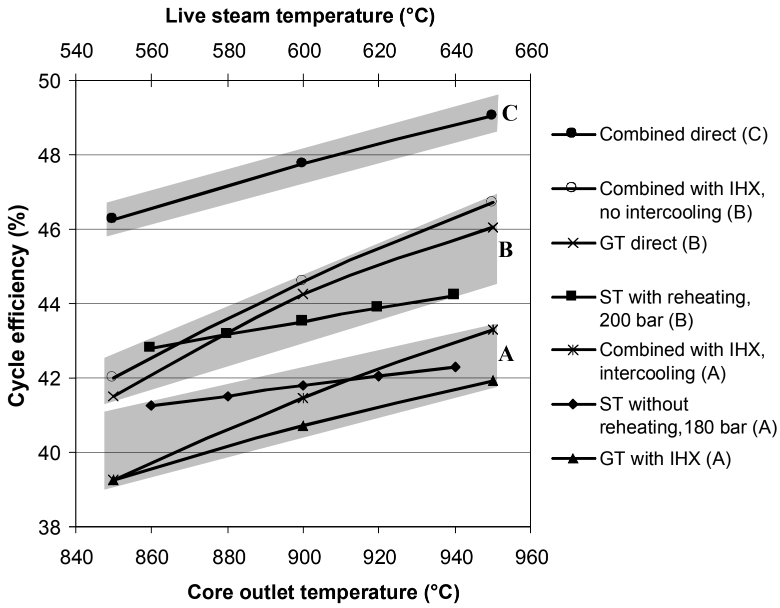

Fig. 6 Overview on efficiencies

Parameter	Steamcycle		Gasturb.c/le		Combined cycle		
	reheater		IHX			IHX	
	no	yes	no	yes	no	yes A	yes B
$P_{nuclear}$, MW	200	200	200	200	200	200	200
$T_{core\ outlet}$, °C	900	900	900	900	900	900	900
$T_{core\ inlet}$, °C	256	256	562	444	286	227	397
P_{blower} , MW	1.7	1.7	_	4.3	_	3.3	4.1
Pressure $Ratio_{total}$	_	_	2.7	4.5	2.5	10	10
$p_{live\ steam}$, bar	160	250	_	_	110	40	40
$T_{live\ steam}$, °C	600	600	_	_	547	408	408
$P_{cycle\ consumption}$, MW	3.5	4.4	0	4.3	0.9	2.5	4.4
P_{net} , MW	82.9	88.2	88.6	81.3	95.0	82.9	89.1
η_{cycle} , %	41.5	44.1	44.3	40.7	47.5	41.5	44.6

A: intercooler between compressor stages, B: no intercooler

Table 1 Main thermodynamic data of the analyzed cycles

5. References

[1] Minatom; DoE; General Atomics; Framatome; Fuji Electric: Executive Summary of Gas-Turbine Modular Helium Reactor Conceptual Design. Framatome, Paris, June 30, 1999

[2] Nicholls, D. R.: Nuclear Engineering International, Dec. 1998, p. 12

[3] Kadak, A. et al.: Modular Pebble Bed Reactor Project, 1st ann. report. MIT, July 1, 1999

[4] HTR-GmbH: THTR-300 MW Kernkraftwerk Hamm-Uentrop, Kurzbeschreibung. Dec. 1983